THEORETICAL ASSESSMENT OF THE FRICTION COEFFICIENT OF A THRUST ROLLER BEARING

ТЕОРЕТИЧНА ОЦЕНКА НА КОЕФИЦИЕНТА НА ТРИЕНЕ НА АКСИАЛЕН РОЛКОВ ЛАГЕР

Assoc. Prof. Dr. Eng. KOLEV P. K.
Technical University of Sofia, Bulgaria

Abstract: *The coefficient of friction of the widely used today rolling bearings is one of the parameters, that influences the efficiency and thermal regime in mechanical equipment and the searching for effective analytical determination is a logical target for research in this area. The complexity of the modern theory of EHL provides an opportunity to seek simpler solutions. In previous research it has found that the classical equation of Reynolds "make difference" between rolling friction and sliding friction and for these two cases and their combination are obtained expressions for the friction force in contact of plate on cylinder in fluid friction. In the present study is derived an expression for the corresponding normal load and are calculated the coefficient of friction as well as the friction torque at selected thrust roller bearing as a function of fluid layer thickness, normal loading and rotational speed. Qualitatively, the corresponding diagrams have confirmation from those published in other studies. Quantitatively, the obtained friction ratio varies from 0,0001 to 0,001, applies to main contact only.*

KEYWORDS: LUBRICATED CONTACT, PLATE ON CYLINDER, REYNOLDS, THRUST ROLLER BEARING, FRICTION COEFFICIENT

1. Introduction

Rolling bearings are machine elements with the most widely application in all fields of technology. One of the main advantages is the low coefficient of friction over a wide range of rotational speeds, whith a lower bound $n_{min}\approx 0$, which is an immanent feature of the physical phenomenon of friction in the dry rolling. In order to reduce the wear, in bearing technique are used plastic or liquid lubricant substances, the presence of which, especially the latter, is a prerequisite for the realization of fluid friction in the contact zones between rolling elements and the paths of the bearing. The liquid friction is more likely in the upper frequency range of the bearing, where $n\approx n_{max}$ and is also characterized whith a low coefficient of friction. The friction coefficient μ define the power losses and thermal regime of the bearing and his determinativeness is an essential prerequisite for the optimal design of power transmissions.

2. Prerequisites for the problem solution

The practical value of the knowing of the friction coefficient is a main reason for the presence of a large number of studies devoted to the determination of its derivative value - the friction torque of the bearing M_R. Contemporary review on this topic is given in [7] and [9]. First Stribeck [2] has formulated the known relation $M_R=\mu.F.d_m/2$. Here F is the load, $d_m=(d+D)/2$ is the average and d and D are the inner and outer diameters of the bearing. Later Palmgren [3] provides the expression $M_R=M_0+M_1$, where M_0 and M_1 are respectively independent and dependent on the load F components of friction torque. This equation is still used by reputed manufacturers of bearings such as INA, FAG and until recently SKF. M_0 depends on the type of bearing and lubrication, on the oil viscosity ν and on the rotational speed n. For values $\nu.n<2000\ mm^2/s.min$, M_0, according to [3], is negligible [9]. For the torque M_1 in [8] and [11] is proposed to place $M_1=f_1.P_1.d_m$. Factor f_1 and workload P_1 are determined according expressions, depending on the type of the bearing. The equations in [2] and [8] are similar and when $M_0\approx 0$ is valid $M_R\approx M_1$. Then, as the meaning $F=P_1$, it turns out $\mu/2\approx f_1$. For μ give [3] and [4] values from 0,001 to 0,005, depending on the type of the bearing - for example, for a thrust bearing with short cylindrical rollers $\mu=0,004$ [9]. For the same axial roller bearing, in [8] and [11] is given the value $f_1=0,0015$. The attitude $\mu/f_1=0,004/0,0015=2,67$ confirms the relationship $\mu/f_1\approx 2$.

Basis for the study of phenomena in the loaded dynamic contact of two solids in the presence of lubrication is the equation of Reynolds [1], which in the case of tangential sliding of cylinder on the plane for isothermal two-dimensional viewing, constant density and constant pressure p in film thickness h_x has the form [7], [10]:

$$\frac{d}{dx}\left(h_x^3.\frac{dp}{dx}\right)=6.\eta.U.\frac{dh_x}{dx} \qquad (1)$$

Here x is the tangential coordinate, η is the dynamic viscosity, $U=U_1+U_2$ is the effective hydrodynamic speed, U_1 and U_2 are the linear velocities of the two bodies in the contact area. The general form of (1) is a basic equation of the hydrodynamic (HD) lubrication theory. The contact of elastic bodies is now a subject of the elastohydrodynamic lubrication (EHL) theory [7]. For example, well known is the isothermal solution for the minimum thickness of lubricant layer in contact of elastic bodies under incompressible Newtonian fluid $h_{min}=2,65.G^{0,54}.U^{0,7}/W^{0,13}$. Here G, U and W are parameters of material, speed and load in the contact. EHL is developed by considering the thermal effects, the properties of the nonnewtonian fluids, the rheology of the lubricants and others. In [7] predicate that "[modern numerical solutions based on EHL, describe well the distribution of pressure, temperature and thickness of the lubricant layer in the contact, but the description of friction is an unsolved problem. Good results are achieved only in the cases when in the contact is present a significant slide. For pure rolling, predominantly in the rolling bearings, it is difficult today to find an adequate solution]". The reason, according to [7] citing [6] is, that "[in heavily loaded bearings, under the influence of extremely high pressure, in the middle of the liquid lubricant layer, the contact surfaces of the two bodies formed glass hardening of the fluid, whose thickness increases with decreasing of the sliding speed and of the heat and that touching in marginal or no sliding. Thus, in pure rolling and rolling with marginal sliding, the friction is no longer carried in the liquid phase, the EHL lubricated contact transform to direct dry contact and hydrodynamic methods are inapplicable]". It is known, however, that in practice the rolling bearings combines rolling and sliding, for example "cinematic" and others, which, especially at low load, make the case described in [7] atypical.

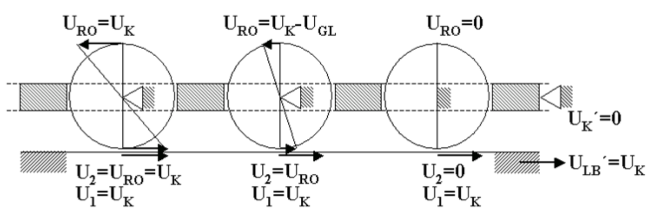

Fig. 1: Types of friction in a rolling bearing [10]

The complexity of the theory of EHL reasons to seek for simpler solutions. In [10], **Fig. 1**, are reviewed the three possible types of friction in the roller bearings, in inverted kinematics: **a**) pure rolling (freely rotating roller), **b**) rolling and sliding (with hampered roller rotation) and **c**) pure sliding (fully blocked roller). With $U_1=U_K$ and $U_2=U_{RO}$ are labeled the peripheral speed of the bearing path and of the roller and through them are expressed the effective hydrodynamic velocity U and the sliding speed U_{GL} in the contact zones for the three cases:

(2)	**a)**	$U=2.U_1$	;	$U_{GL}=0,$
(3)	**b)**	$U=U_1+U_2$	;	$U_{GL}=U_1-U_2$
(4)	**c)**	$U=U_1$	;	$U_{GL}=U_1$

Thus, in [10] is found, that the equation of Reynolds [1] "make difference" between the three types of friction and on this basis is formulated the hypothesis, that this equation is applicable not only to sliding but also to rolling bearings with hydrodynamic lubrication.

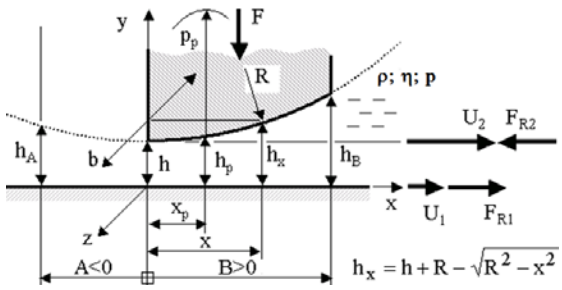

Fig. 2: Model of the contact in the rolling bearing [10]

For description of the contact, in [10] is composed the model shown in **Fig. 2** and are derived dependencies for the pressure p, for the profile of the speed u and for the frictional forces F_R in the lubricant layer for all the three studied cases.

3. Aim of the work

Aim of the work is, based on the findings in [10], to perform a theoretical determination of the friction coefficient and by comparison with previously published values to assess the applicability of equation (1) of Reynolds [1] for the case of rolling bearing at fluid friction.

4. Solution of the problem

The coefficient of friction in the considered contact is defined by the well-known relation $\mu=F_R/F$ between the normal load F and the friction force F_R, shown in **Fig. 2**. Unto given model parameters, the friction forces are defined here on the basis of summary record of the equations derived in [10], namely:

(5) $$I_{FR(I2)}=\frac{1}{a}.arctg\frac{B}{a};\ I_{FR(I1)}=\frac{1}{2.a^2}.\left(\frac{B}{X_B}+I_{FR(I2)}\right);$$

(6) $$F_{R1/2}=\pm 6.\eta.b.U.R.\left[2.R.h_p.I_{FR(I1)}-\left(1\pm\frac{1}{3}.\frac{U_{GL}}{U}\right).I_{FR(I2)}\right]$$

The phenomenon $F_{R1}\neq F_{R2}$ is described in [5], and the conclusion (6) also confirms this. The normal force F is balanced by the pressure p in the lubricant layer, which expresses the load capacity of a part of an infinite cylinder with width b and is obtained here in the form:

(7) $$F=b.\int_0^B p(x).dx \quad \Rightarrow$$

(8) $$C_{F1}=3.\eta.U/h^2;\ C_{F2}=-3.h_p/(4.h);$$

(9) $$I_{F1}=(\ln X_B-\ln a^2)/2;$$

(10) $$I_{F2}=B.arctg(B/a)+a.(\ln a^2-\ln X_B)/2;$$

(11) $$I_{F3}=(a^{-2}-X_B{}^{-1})/2;$$

(12) $$F=b.[\ C_{F1}.(I_{F1}+I_{F2}/a)+ \\ +C_{F2}.(I_{F1}+I_{F2}/a+2.a^2.I_{F3}/3)\ ]$$

The model parameters a, B, X_B and h_p are defined in [10]. As object of study is selected an axial roller bearing 81118 TN from the product catalog of the company SKF [12], **Fig. 3**, whith the geometry of the contact near to the model in **Fig. 2**. The width of the roller is $b=(117\text{-}93)/2=12\ mm$, **Fig. 3**. The upper limit of the integral in (7) is the abscissa $x=B$, **Fig. 2**, where the pressure p in the lubricated layer becomes negligible. Here, according to [10], it is adopting $B\approx R/3$. The referent and the maximum rotational speed of the bearing are 1300 min^{-1} and 2600 min^{-1} [12]. For the operating speed and the operating temperature is taken $n=1300\ min^{-1}$ and $T=70\,°C$, which unto average diameter $(d+D)/2=105\ mm$ defines a minimum kinematic viscosity to provide a liquid friction $\nu_1=10\ mm^2/s$ at 70° C and $\nu=28{,}4\ mm^2/s$ at 40 °C [12]. It is selected an oil ISO class VG100 with $\nu=100\ mm^2/s$ at 40° C and $\nu\approx 27{,}5\ mm^2/s$ at 70° C.

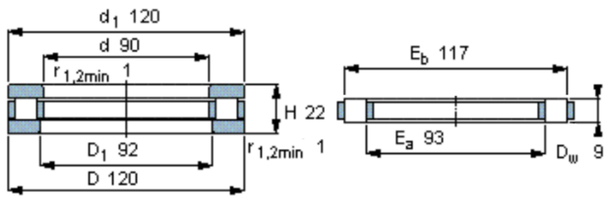

Fig. 3: Thrust roller bearing 81118 TN [12]

Typical for the axial roller bearings is the presence of a cinematic conditioned slide. The velocities plan of the relative movement in case "stopped cage" is shown in **Fig. 4**. $n_1=650\ min^{-1}$ and the bearing size determines the average peripheral speed $U_1=(U_{1a}+U_{1b})/2=3573\ mm/s$ on the bearing paths. For the peripheral roller speed is taken $U_2=U_{1a}=3165\ mm/s$, which determines a sliding speed in the contact $U_{GL}=408\ mm/s$.

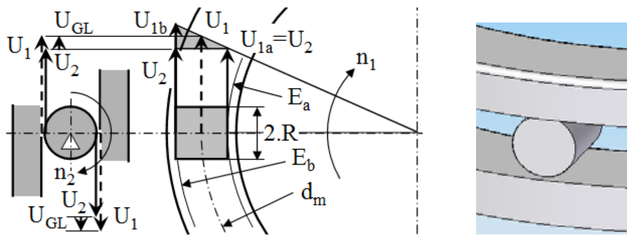

Fig. 4: Plan of velocities in the contact zones

As is meant, that one bearing roller contacts with two bearing washers, **Fig. 4**, that $d_m=(E_a+E_b)/2$ is the average diameter of each of two bearing paths and that the bearing can be seen as a friction gear, **Fig. 5**, for the friction torque M_R, caused in the both contacts of one bearing roller follows:

(13) $$i=\frac{n_1}{n_2}=\frac{2.R}{d_m}=\frac{2.R.F_R}{M_R} \quad \text{and} \quad M_R=\mu.F.d_m$$

In the expressions from (6) to (13) F_R, F, μ and M_R are functions of the thickness of the lubricant layer h, **Fig. 2**. On this basis, after entering of the described numerical values of model parameters, for series of values of the layer thickness h, in the working environment of MS EXCEL are obtained numerical values of the power characteristics and of the friction coefficient in both hydrodynamic contacts of one single roller of the testing bearing.

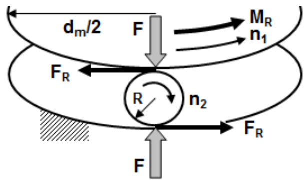

Fig. 5: The axial roller bearing seen as a friction gear

5. Results and discussion

The results of the computational test of the expressions from (6) to (13) are presented here in the form of a graphical correlations for the three cases of friction in the bearing **a**), **b**) and **c**). In the studied interval of the lubricant layer thickness, with decreasing of ***h*** increases both the load capacity of hydrodynamic contact ***F***, **Fig. 6**, and the maximum friction force in the layer $\boldsymbol{F_R}$, **Fig. 7**. These dependencies are formulated in [5] as well, which confirms the principal correctness of the results obtained here.

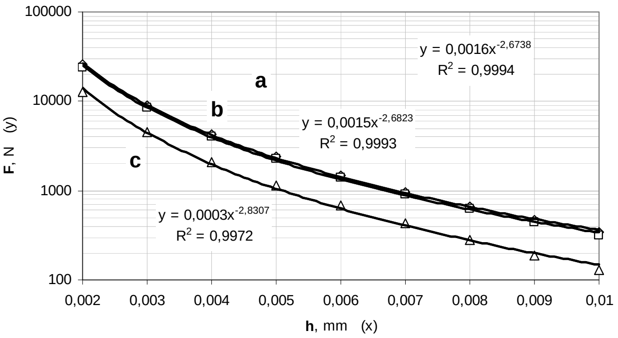

Fig. 6: Layer thickness ***h*** and load capacity ***F***

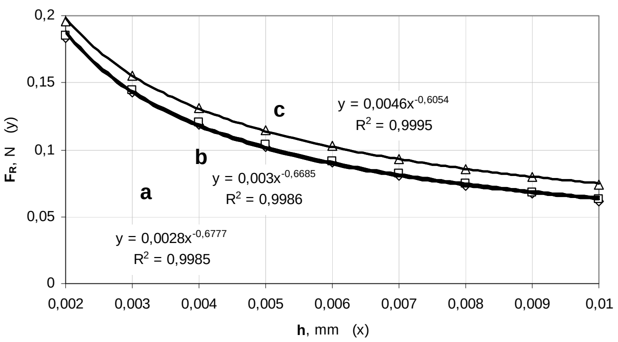

Fig. 7: Layer thickness ***h*** and friction force $\boldsymbol{F_R}$

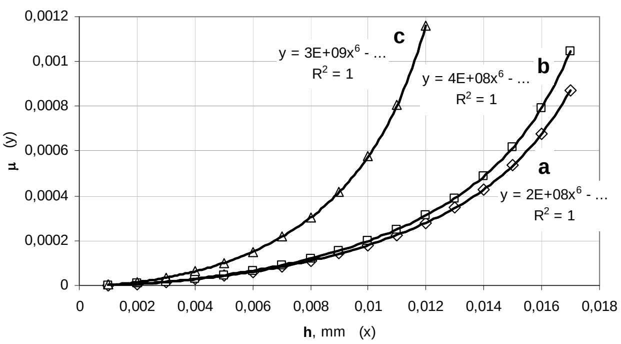

Fig. 8: Layer thickness ***h*** and friction coefficient $\boldsymbol{\mu}$

The maximum ($\boldsymbol{F_{R1}>F_{R2}}$) friction coefficient μ decreases with decreasing of the film thickness ***h***, **Fig. 8**, i.e. parallel with the increasing of the load capacity ***F*** of the fluid friction contact, **Fig. 9**. This principle dependence is also known subject and here is yet another confirmation. The calculated low values of the friction coefficient μ, **Fig. 8**, are typical for the hydrodynamic lubrication at all, which in turn speaks for the correctness of the conclusions from (5) to (12). **Fig. 8** shows, how these equations, respectively the equation of Reynolds [1], report the influence of sliding in the contact, giving significantly different results in the ideal case **a**) for the research bearing, namely the pure rolling, and practically the unlikely but possible case **c**), when all the rollers are blocked, not rotate and the bearing does not work as a rolling bearing, but as axial plain bearing.

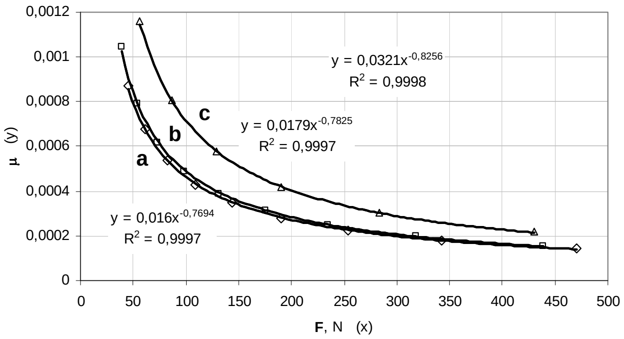

Fig. 9: Load capacity ***F*** and friction coefficient $\boldsymbol{\mu}$

Fig. 8 leads to the conclusion, that unto the fluid friction the pure rolling **a**) is an operating mode, energetically more favorable in comparison with the case of pure sliding **c**). The real operating mode **b**) in the research thrust bearing with short cylindrical rollers is predominantly pure rolling accompanied by minor cinematic sliding and is close to the perfect mode **a**) of the pure rolling unto the fluid friction and therefore the graphs describing the cases **a**) and **b**) are close to each other and nearly overlap.

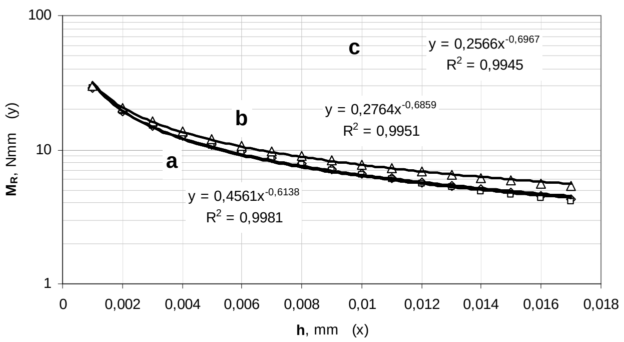

Fig. 10: Layer thickness ***h*** and friction torque $\boldsymbol{M_R}$

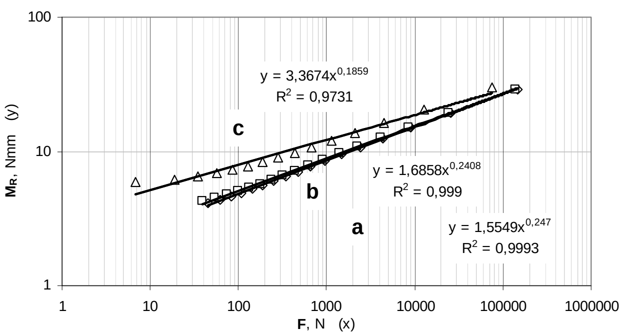

Fig. 11: Load capacity ***F*** and friction torque $\boldsymbol{M_R}$

The values of the friction coefficient, obtained here for one single hydrodynamic contact, $\mu\approx 0,0001\text{-}0,001$, **Fig. 8**, are up to an order of magnitude lower than the indicated in [9] value $\mu=0,004$

for a complete thrust roller bearing. The main reason is the simple two-dimensional model of the contact, **Fig. 2** [10], which is limited only in the half-plane $x \geq 0$. Furthermore, the conclusions and calculations made here for the main contact in the bearing, between the roller and the bearing washers unto fluid friction, not cover the contact friction between the cylindrical and top surfaces of the roller on the walls of the cage window, that contributes to an increase in the coefficient of friction.

The test calculation shows, that the friction torque M_R, caused by contact of a single roller with the two washers increases when the fluid layer thickness h reduces, **Fig. 10** and when the load F increases, **Fig. 11**. The latter dependence is in agreement with the results in [7] and is in agreement with the quoted above equation of Palmgren [3], that contains a component of the friction torque M_1, proportional to the load F.

Here is also done a numerically study of the impact of the roller rotational speed n_1, relatively to the bearing washers, on the friction coefficient μ, **Fig. 12** and on the friction torque M_R, **Fig. 13**., when the lubricant film thickness is taken $h=8\ \mu m$.

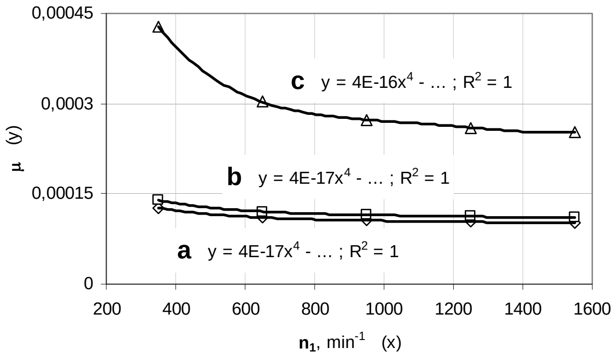

Fig. 12: Rotational speed n_1 and friction coefficient μ

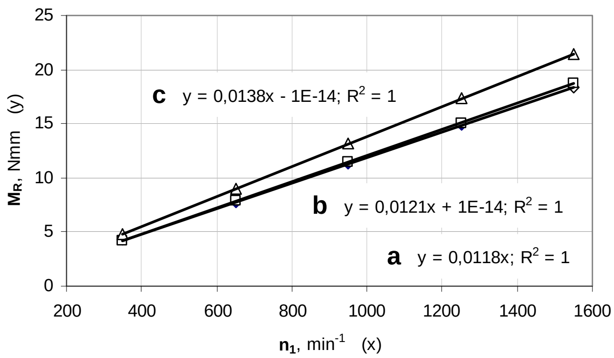

Fig. 13: Rotational speed n_1 and friction torque M_R

The both diagrams in **Fig. 12** and in **Fig. 13** are generally confirmed by the published experimental results in the doctoral theses [7] and [9], for example, and **Fig. 12** shows the apparent greater stability of the friction coefficient μ in case **a)** and **b)**, where the pure rolling prevails.

6. Conclusion

The friction coefficient of the mass used now rolling bearings is one of the parameters that affect the efficiency and thermal regime in machine technology and the search for effective method for his analytical determination is a logical target for research in this area. The complexity of the modern theory of EHL reasons to seek simpler solutions. It is found in a previous research, that the classical equation of Reynolds "make difference" between rolling friction and sliding friction and for these two cases and their combination are obtained expressions for the friction force in the contact of plate on cylinder in fluid friction. In this study is derived an expression for the corresponding normal force and are calculated the friction coefficient and the friction torque in a selected thrust roller bearing, as a function of the lubricant layer thickness, of the load and of the rotational speed. The here obtained graphical relationships have received confirmation from those, published in other studies, qualitatively. Quantitatively, the values obtained for the coefficient of friction, from 0,0001 to 0,001, only for the main contact between the bearing roller and path, are up to an order of magnitude lower than indicated in the literature, eg 0,004 for a complete bearing. The difference would be smaller, when one considers the friction in other contact pairs of the bearing, which is not addressed here. The obtained results are initial and can serve as a basis for further theoretical and experimental studies.

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